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State-of-the-art of solar thermal power plants—A review

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ABSTRACT

The solar thermal power plant is one of the promising renewable energy options to substitute the increasing demand of conventional energy. The cost per kW of solar power is higher and the overall efficiency of the system is lower. In the present communication, a comprehensive literature review on the scenario of solar thermal power plants and its up-to-date technologies all over the world is presented. Results of the technical and economical feasibility studies by researchers are reported in brief for further reference. It is observed that the solar thermal power plants have come out of the experimental stage to commercial applications. Case studies of typical 50 MW solar thermal power plants in the Indian climatic conditions at locations such as Jodhpur and Delhi is highlighted with the help of techno-economic model. Different solar concentrator technologies (parabolic trough, parabolic dish and central power tower) for solar thermal power plants are compared economically. It has been found that the parabolic dish concentrating solar Stirling engine power plant generate electricity at a lower unit cost than the other two solar technologies considering 30 years lifespan and 10% interest rate on investment.

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1. Introduction

The ever increasing demand of energy for development of the society is fulfilled by a variety of energy sources. Large scale energy utilization has led to a better quality of life and faster all round development; it has also generated many critical problems [1]. The most prominent of these is the harmful effect on the environment in various forms leading to global warming and climate change [2]. At the same time, the fossil fuel resources are also fast depleting due to over exploitation. Therefore, it is worth to explore the alternative energy sources, systems and technologies for sustainable development, if not fully but at least to substitute an appreciable amount of conventional energy to mitigate the harmful effect to some extent.

Other than fossil fuels, nuclear and large hydro-power, there are a number of sources of energy which have started contribution in a small way to the world's present energy demand and supply scenarios. These include energy sources like wind energy, small hydro, photovoltaic conversion, bio-mass, tidal, geothermal energy and solar thermal power plants. Among the renewable energy

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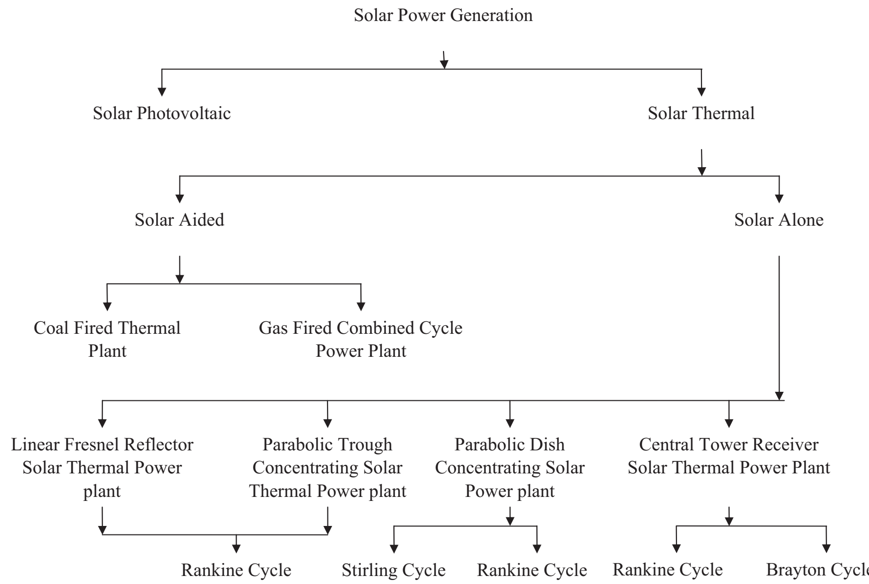


Fig. 1. Schematic view of solar power generation methods.

sources, solar power generation undoubtedly offers the most promising and viable option for electricity generation for the present and future. The schematic views of solar power generation methods are shown in Fig. 1.

In the present study, the authors have focused on the solar thermal conversion route of power generation only. The basic mechanism of conversion and utilization of solar energy for solar thermal power generation is available in the literature elsewhere. The main differences are found to be in the solar energy collection devices, working fluids, solar thermal energy storage and heat-exchanger, and suitable solar thermal power cycles. Solar thermal power cycles are classified as low (up to 100° C), medium (up to 400° C) and high (above 400° C) temperature cycles [1].

2. Status of low and medium temperature technologies of solar thermal power plants

Low temperature solar thermal power plants use flat-plate collectors, or solar ponds for collection of solar energy. The working fluid of low boiling points; organic fluids like methyl chloride and toluene, and refrigerants like R-11, R-113 and R-114 are normally used in the Rankine cycle. Solar power plants of this type having generation capacities up to about 50 kW were installed in many parts of the world, particularly Africa, in 1970s. The reported Rankine cycle efficiency of 7–8% and efficiency of the solar flat-plate collector system of about 25% lead to an overall efficiency of only 2%. The cost of similar solar thermal power plant of 10 kW installed at IIT Chennai in 1979–1980 were estimated about Rs. 300,000 per kW for 6–8 h of daily operation. In order to reduce the cost, solar ponds have been used instead of flat-plate collectors in Israel for 6 kW and 150 kW capacities. Systems working on the solar chimney concept have also been tried in Manzanares, Spain as an experimental pilot plant [1].

Medium temperature solar power plants use the line focusing parabolic solar collector at a temperature about 400° C. Significant advances have been made in parabolic collector technology as well as organic Rankine cycle technology to improve the performance of parabolic trough concentrating solar thermal power plant (PTCSTPP). A parabolic trough collector consists of long parallel rows of reflectors made by bending a sheet of reflective material (silvered low-iron float glass) into a parabolic shape [3]. At the focal point of the reflector is the absorber tube or receiver. The receiver is a black treated metal tube, covered with a glass tube, the space between the pipe and glass cover is evacuated to reduce heat losses. The rows are arranged along a north–south axis and they can rotate from east to west over each day. Parabolic troughs can achieve concentration ratios (ratio of solar flux on the receiver to that on the mirrors) of between 10 and 100. A heat transfer fluid (HTF) is circulated through the receiver absorber tube to remove the solar heat. The HTF can be heated to temperatures of up to 673 K. The fluid is pumped to a heat exchanger where its heat is transferred to water or steam.

A tracking mechanism is to be used to follow the sun and must be able to track the sun during periods of intermittent cloud cover. Finally, then return the parabolic trough concentrator to its original position at the end of the day or during the night. At present tracking system for the parabolic trough concentrator is based on “virtual” tracking. The traditional sun-tracking unit with sensors that detect the position of the sun has been replaced by a system based on calculation of the sun position using a mathematical algorithm [4]. The biggest application of this type of system is the southern California power plants known as Solar Electric Generating Systems (SEGS), which have a total installed capacity of 354 MW_e [5]. SEGS-I of 14 MW_e capacity was set up in 1984. SEGS-II to VII was of 30 MW_e each, and SEGS-VIII and IX are of 80 MW_e each. The collector array for SEGS-IX has an area of 483,960 m^2. SEGS-VIII which started operation in 1990 is reported to have cost $4000 per kW.

A recent development in cost effective concentrators is the design of the Euro Trough, a new parabolic trough concentrator, in which an advanced lightweight structure is used to achieve cost-efficient solar power generation [6,7]. More details on this development of parabolic concentrator system are given in Table 1.

The parabolic trough solar power plant can collect up to 60–70% of the incident solar radiation and has achieved a peak electrical conversion efficiency of 20–25% (net electricity generation to incident solar radiation). A number of researchers have carried out works on the solar parabolic trough concentrator based solar power system with varied perspectives. Some of the findings are presented in this paper for further reference in the research works with the aim to improve the performance of the solar thermal power generation systems. Treadwell et al. [8] presented the performance of single-axis tracking parabolic trough solar

Table 1
Data on one-axis parabolic trough concentrators [3].

Collector	Structure	Aperture width (m)	Focal length (m)	Length per element (m^2)	Length per collector (m)	Mirror area per drive (m^2)	Receiver diameter (m)	Geometric concentration (SU)	Mirror type	Drive	Peak optical efficiency (%)
LS-1	Torque tube	2.55	0.94	6.3	50.2	128	0.04	61:1	Silvered low-iron float glass	Gear	71
LS-2	Torque tube	5	1.49	8	49	235	0.07	71:1	Silvered low-iron float glass	Gear	76
LS-3	V-truss framework	5.76	1.1	12	99	545	0.07	82:1	Silvered low-iron float glass	Hydraulic	80
New IST	Space frame	2.3	0.76	6.1	49	424	0.04	50:1	Silvered thin glass	Jack screw	78
Euro trough	Square truss torque box	5.76	1.71	12	150	817	0.07	82:1	Silvered low-iron float glass	Hydraulic	80
Duke solar	Aluminum space frame	5	1.49	8	49–65	235–313	0.07	71:1	Silvered low-iron float glass	Hydraulic or gear	80

collectors based on the typical meteorological year input data of 11 sites. North–south horizontal axis parabolic trough collector performance superiority has been recommended based on their results. Treadwell and Grandjean [9] discussed the certain systematic errors in the angle between the reflector vertex-focus axis and the vertex-sun axis and systematic receiver location error in the vectorial deviation of a receiver from focus on performance and, therefore, their influence on the design of troughs. Guven and Bannerot [10] presented the mathematical derivation of concentration ratio and rim angle in parabolic troughs. Guven et al. [11] presented a rational approach for multi-objective design and optimization of parabolic trough solar collectors for different design environments. Odeh and Morrison [12] developed a transient simulation model for analysis of the performance of industrial water heating systems using parabolic trough solar collectors. Tao et al. [13] presented the operational principle and design method of a new trough solar concentrator. The influence of important design and characteristic parameters are analyzed and optimized in the paper.

Mullick and Nanda [14] presented a different approach to evaluate the heat loss factor of a tubular absorber with a concentric glass cover. Kearney et al. [15] carried out the feasibility of utilizing a molten salt as the heat transfer fluid and for thermal storage in a parabolic trough solar field to improve system performance. Bakos et al. [16] developed a simulation program, based upon the variation of collector's efficiency as a function of heat transfer fluid flux, pipe diameter, solar radiation intensity and active area of the parabolic trough concentrator. Naeeni and Yaghoubi [17] presented a study on heat transfer from a receiver tube of the parabolic trough collector of the 250 kW solar power plants in Shiraz (Iran). The effects of variation of collector angle based on wind velocity are studied here.

You and Hu et al. [18] studied feasibility of the reheat-regenerative Rankine power cycle for the parabolic trough collector. They also investigated the optimal thermal and exergetic efficiencies for the collector and power cycle. Gang et al. [19] designed a low temperature solar thermal electric generation with and without regenerative heat exchanger in organic Rankine cycle (ORC). It has been found that the ORC with regenerative heat exchanger has the efficiency of about 8.6% which is relatively higher by 4.9% than that without the regenerative heat exchanger. Fernandez-Garcia et al. [20] presented a survey of concentrating solar system especially for the steam power cycles for electricity generation.

Kerkeni et al. [21] presented a close analysis and evaluation of the long-term performance of the system. Lechon et al. [22] discussed the opportunities to improve the performance of 50 MW solar parabolic trough thermal plants in Spain, by the life cycle assessment in order to reduce their environmental impacts. Montes et al. [23] described the influence of the solar multiple on the annual performance of parabolic trough solar thermal power plants with direct steam generation. Munoz et al. [24] proposed a conceptual design of a solar boiler and found that overall efficiency of the conversion of direct solar irradiation energy to electricity is above 20%. Mohammed et al. [25] proposed and analyzed a prototype of a 50 MW concentrated solar power plant (CSPP) based on the solar irradiation data for electricity generation in Jordan. It was found that Jordan has an outstanding potential for CSPP, especially in the southern locations of the country. Birnbaum et al. [26] have done comparative analysis of direct steam generation parabolic trough power plants with and without thermal storage facility. They found that depending on the live steam parameters, a reheat is necessary within the power block. Yan et al. [27] presented a dynamic model of solar parabolic trough collectors using explicit Euler's method. Different working conditions of the collector structure and thermal parameters have been considered in this model. The simulated results are validated using the selected real test data on typical summer and winter days. Garcia et al. [28] described a simulation model that predicted the performance of parabolic trough solar thermal power plants with a thermal storage system. Results based on this model of a 50 MW power plant are presented and compared to the real performance data.

Montes et al. [29] analyzed the contribution of solar thermal power to improve the performance of gas-fired combined cycles. An integrated solar combined cycle power plant was proposed which consists of a parabolic trough field coupled to the bottoming steam cycle of a combined cycle gas turbine power plant. Garcia-Barberena et al. [30] developed SimulCET computer program for analysis of the influence of different operational strategies on the performance of parabolic trough solar power plants. The results generated by SimulCET were validated with current experimental data. Feldhoff et al. [31] described and compared the two types of plants (direct steam generated system with and without thermal energy storage) based on their design & performance. The results indicated further effort in the development of a commercial storage system for direct steam generated solar power plants. Bonilla et al. [32] developed a dynamic simulation for design and development of a direct steam generation parabolic trough solar thermal power plant. The dynamic simulation is not only the equation-based object-oriented model but also includes features to facilitate the simulation process.

Kreider [33] explained entropy level of the solar resource as converted to heat in various types of solar collectors. Singh and Kaushik [34] analyzed solar thermal power system using finite-time thermodynamics in order to find the optimum operating

temperature. Singh et al. [35] performed energy analysis based on exergy concept of second law of thermodynamics for a solar thermal power system. Basic energy and exergy analysis for the system components (viz parabolic trough collector/receiver and Rankine heat engine, etc.) are carried out for evaluation of the respective losses as well as exergetic efficiency for typical solar thermal power systems under given operating conditions.

Gupta and Kaushik [36] analyzed the possibilities of further improvement in efficiency of the solar thermal power plant (STPP) and evaluated the optimum bleed pressure and mass fraction of bleed steam for the enhancement of the efficiency of the solar thermal power plant. Montes et al. [37] described a thermo fluid dynamic model for parabolic trough collectors. Based on this model they analyzed the influence of factors (heat loss and pressure drop) for energy, and exergy efficiencies with different working fluids: oil, molten salt, or water/steam. Siva Reddy et al. [38] have evaluated the energetic and exergetic losses as well as efficiencies for typical parabolic trough concentrating solar thermal power plant (PTCSTPP) under the specific operating conditions. It is found that by increasing the operating pressures of the solar thermal power plant (STPP) from 90 bar to 105 bar pressure, the energetic and exergetic efficiencies of PTCSTPP are increased by 1.49% and 1.51%, respectively.

The above mentioned works on the solar thermal power plants based on parabolic trough concentrator technology prove technical viability of its operation and provide a strong base for further research and development in this area of future need of sustainable development. At the same time, economic viability must also be judged for the adoption of any new technology by the society. Therefore, the value of the solar thermal power generation must ultimately be judged in economic terms.

Power generation options are most commonly compared on the basis of their unit electricity (kWh) costs. The total cost comprises of initial capital investment and annual operating and maintenance costs over the useful life of the plant. It is pertinent to mention here that prediction of unit electricity costs for new technologies are subject to many uncertainties that significantly influence whether or not the component projections for capital cost, annual performance, and operation & maintenance cost are met. These uncertainties include system reliability, equipment efficiencies & lifetimes, organizational learning, manufacturing capability, and technological improvements. Many researchers have studied about the economic analysis of solar thermal power plants with parabolic trough collector. It is worth to be mentioned here for further reference by researchers worldwide.

Gee and Murphy [39] presented economic analysis by using the performance potential of selected parabolic through component improvements. Upper bound costs for each improvement were estimated, and they concluded increased solar energy systems rates of return made possible by these improvements Luzzi et al. [40] estimated levelized electricity costs of solar thermal power on a continuous, 24-h operation by using thermo chemical energy storage. The levelized electricity costs of less than 0.26$/kWh has been estimated with a net solar-to-electric conversion efficiency of 18% and a capacity factor of 80%. Quaschning et al. [41] proposed a new method for estimating the optimization of solar field size as a function of the solar irradiance and economic aspects using smartest simulation tool.

Horn et al. [42] studied feasibility of an integrated solar combined cycle system (ISCCS) for both technical and economical viability in Egypt with support from the global environment facility. Both, parabolic trough collector field and volumetric air receiver tower were considered as possible solar systems. They found the levelized electricity cost for ISCCS was 3.1$/kWh. At the same time, solar system levelized electricity cost was 9.5$/kWh. Hosseini et al. [43] performed technical and economic assessment of solar power plants based on the main parameters like thermal efficiency, capacity factor, environmental considerations, investment, fuel and O&M costs.

Poullikkas [44] carried a feasibility study in order to investigate whether the installation of a parabolic trough solar thermal technology for power generation in the Mediterranean region is economically viable or not. Laing et al. [45] analyzed option of solid media sensible heat storage for parabolic trough power plants using synthetic oil as the heat transfer medium in terms of investment and maintenance costs. They found a decrease in levelized energy costs with a modular storage integration of 2–3%. Vallentin and Viebahn [46] analyzed possible value creation effects resulting from a global deployment of CSP until 2050 as projected in scenarios of the International Energy Agency (IEA) and Greenpeace International.

Feldhoff et al. [47] investigated economic feasibility of the direct steam generation (DSG) parabolic trough collectors to improve the mature parabolic trough solar thermal power plant technology of the solar energy generating systems in California. The main result of the investigation is to show that the levelized electricity cost reduction can obtain up to 11% based on the tropical condition. Purohit [48] analyzed the financial feasibility of CSP technologies in Indian conditions with reference to two projects namely PS-10 (based on power tower technology) and ANDASOL-1 (based on parabolic trough technology). It is reported that the possibility of success of these technologies in the north-western part of the country are more especially in Rajasthan and Gujarat states.

Poullikkas et al. [49] presented technical and economic analysis for the integration of parabolic trough concentrated solar power technologies, with or without thermal storage capability of 50 MW or 100 MW capacities. Sharaf et al. [50] analyzed and evaluated electrical power generation from the solar organic Rankine cycle with parabolic trough collector using toluene organic oil, Water and Therminol-VP1 working fluids, thermo-economically. Spelling et al. [51] proposed a dynamic model for multi-objective thermo economic optimization of both the power plant performance and cost, using a population-based evolutionary algorithm. Zamfirescu et al. [52] analyzed the exergy interactions, environmental impact in terms of CO_2 mitigation, and the economics of small-capacity concentrated solar power-driven heat engines for power and heat generation for residential applications. Nixon and Davies [53] presented a new method for the optimization of the mirror element spacing arrangement and operating temperature of linear Fresnel reflectors (LFR) for maximizing available power output and minimizing the cost.

3. Status of high temperature technologies of solar thermal power plants

Two types of concentrator systems: the paraboloid dish-Stirling engine and the central tower receiver are primarily tried for high temperature solar thermal power plants in the world. The paraboloid dish concentrator-Stirling engine solar thermal power plants (PDCSSPP) developed for commercial applications generate power in kW and found to be suitable for power supply in communities and villages especially in rural areas. The central tower receiver solar thermal power plants (CTRSTPP) are capable of generating electrical power in MWs.

In PDCSSPP, the paraboloid dish concentrator tracks the sun by rotating about two axes and the incident sun-rays are focused on a point. Complete two axis tracking of the concentrator aperture would increase the amount of insolation incident on by elimination of the cosine effect [54]. At focal point the absorber in the Stirling receiver absorbs solar radiation and transfers the thermal energy to the Stirling engine. Current Stirling absorbers are

typically direct illumination receivers and heat pipe receivers and volumetric receivers.

The heat pipe absorbers vaporize a liquid metal such as sodium on the absorber surface and condense it on the Stirling engine heater tubes to transfer the energy to the working fluid. Heat pipe receivers yield more uniform temperature distributions on the heater tubes; thereby, resulting in longer life for the absorbers and engine heater heads in comparison to the direct illumination receiver absorbers. Volumetric receivers have the potential to be more cost effective and reliable than the heat pipe absorbers and are used in hybrid natural gas Stirling dish systems [55]. Table 2 shows the detailed of the specifications and performance parameters of the four dish-Stirling systems, currently in use on commercial scale.

A numbers of research works on the parabolic dish concentrator and Stirling engine system is reported in literature. Clausing [56] presented an analytical model for the estimation of convective heat losses from cavity receivers. Significant convective heat losses from cavity receivers are indicated in this study. Bannister [57] examined the problem of optimizing the radius of boiler tubes in a radiation-dominated environment such as parabolic dish solar thermal collector–receiver. Chen et al. [58] investigated the performance of the system based on the linearized heat loss model of the solar collector and the irreversible cycle model of the Stirling engine. Optimal operating temperature of the solar collector at the maximum efficiency of the system was determined.

Kaushika and Reddy [59] presented the design, development and performance characteristics of a low cost solar steam generating system which incorporates recent design and materials innovations of parabolic dish technology. Sendhil Kumar and Reddy [60] presented a numerical investigation to study the natural convective heat loss from three types of receivers (cavity receiver, semi-cavity receiver and modified cavity receiver) for a fuzzy focal solar dish concentrator.

Reddy and Kumar [61] also presented the numerical study of combined laminar natural convection and surface radiation heat transfer in a modified cavity receiver of solar parabolic dish collector. A two-dimensional simulation model for combined natural convection and surface radiation was developed. The influence of operating temperature, emissivity of the surface, orientation and the geometry on the total heat loss from the receiver is investigated. The convective heat loss from the modified receiver is significantly influenced by the inclination of the receiver whereas the radiation heat loss is considerably affected by surface properties of the receiver. A three-dimensional simulation model has also been developed to investigate the accurate estimation of natural convection heat loss from modified cavity receiver without insulation and a comparison of 2-D and 3-D natural convection heat loss from a modified cavity receiver is carried out. Reddy and Kumar [62] found that the 3-D model can be used for accurate estimation of heat losses from solar dish collector, when compared with other well known models.

Prakash et al. [63] carried an experimental and numerical study of the steady state convective losses occurring from a downward facing cylindrical cavity receiver and developed correlations for certain receiver geometries. Wu et al. [64] proposed a parabolic dish/alkali metal thermal to electric converter (AMTEC) solar thermal power system and evaluated its overall thermal electric conversion performance. Results show that the overall conversion efficiency of parabolic dish/AMTEC system could reach up to 20.6%. A comprehensive review and systematic summarization of the research progress in the parabolic dish concentrator presented by Wu et al. [65] is worth to be appreciated. Lovegrove et al. [66] had given the new design of a 500 m^2 concentrator with 13.4 m focal length and altitude-azimuth tracking paraboloidal dish concentrator.

Li and Dubowsky [67] presented an analytical model to optimize the shape and thickness of the petals. The concept is demonstrated using Finite Element Analysis and laboratory experiments. The Monte-Carlo ray-tracing method is utilized to predict the radiation flux distributions of the concentrator receiver system

Table 2
Comparative specifications and performance parameters for parabolic dish concentrator solar Stirling engine systems [55].

Parameters	SAICOSTM system	SBP system	SES system	WGA (Mod 1) ADDS system	WGA (Mod 2) remote system
Concentrator					
No. of facets glass	16	12	82	32	24
Area (m^2)	117.2	60	91.0	42.9	42.9
Projected area (m^2)	113.5	56.7	87.7	41.2	41.2
Reflectivity	0.95	0.94	0.91	0.94	0.94
Height (m)	15.0	10.1	11.9	8.8	8.8
Width (m)	14.8	10.4	11.3	8.8	8.8
Focal length (m)	12.0	4.5	7.45	5.45	5.45
Intercept factor	0.90	0.93	0.97	0.99	0.99
Peak CR (SU)	2500	12,730	7500	11,000	13,000
Power conversion unit					
Aperture diameter (cm)	38	15	20	14	14
Engine manf/type	STM 4-120 double acting kinematic	SOLO 161 kinematic	Kockums/SES 4-95 kinematic	SOLO 161 kinematic	SOLO 161 kinematic
No. of cylinders	4	2	4	2	2
Displacement (cm^3)	480	160	380	160	160
Operating speed (rpm)	2200	1500	1800	1800	800–1890
Working fluid	Hydrogen	Helium	Hydrogen	Hydrogen	Hydrogen
Power control	Variable stroke	Variable pressure	Variable pressure	Variable pressure	Variable pressure
Generator	3Φ/480v/Induct	3Φ/480v/Induc	3Φ/480v/Induct	3Φ/480v/Induc	3Φ/480v/synch
System information					
No. of systems built	5	11	5	1	1
Rated output (kW)	22	10	25	9.5	8
Peak output (kW)	22.9	8.5	25.3	11.0	8
Net peak efficiency (%)	20	19	29.4	24.5	22.5

for uniform heater temperature and high optical thermal efficiency of a Solar Dish/Stirling engine [68]. Wu et al. [69] performed an optimal performance analysis for a Stirling engine with heat transfer and imperfect regeneration irreversibilities and derived a relation between the net power output and thermal efficiency. Senft et al. [70] described a mathematical model of engines operating with an ideal Stirling cycle and subject to limited heat transfer/internal thermal losses and mechanical friction losses and analyzed the fundamental effects of these imperfections on the performance of an ideal Stirling engine. Costea et al. [71] studied the effect of pressure losses and actual heat transfer on the performance of a solar Stirling engine. Pressure losses, due to fluid friction internal to the engine and mechanical friction between the moving parts, were also estimated through extensive and rigorous use of the available experimental data.

Berrin Erbay and Yavuz [72] evaluated theoretically, the effects of inefficiencies in the compression, expansion and regeneration processes on engine performance. The irreversible cycle has been optimized by using the maximum power density technique. Bhattacharyya and Blanks [73] evaluated a major theoretical consideration concerning the design of an endoreversible Stirling cycle with ideal regeneration. The factors affecting optimum power and efficiency at optimum power are analyzed for the cycle based upon higher and lower temperature bounds. Senft [74] presented combined mechanical efficiency of reciprocating engines with the classic Schmidt thermodynamic model for Stirling engines for identifying optimal engine geometry.

Kaushik et al. [75] presented the performance evaluation of irreversible Stirling and Ericsson heat pumps cycles including external and internal irreversibilities along with finite heat capacities of external reservoirs. Petrescu et al. [76] developed a model based on the first law of Thermodynamics for calculating the efficiency and power of Stirling machines. Kongtragool and Wongwises [77] presented a review upon the development of Stirling engines and solar-powered Stirling engines. Timoumi et al. [78] developed a numerical simulation model to investigate the influence of geometrical and physical parameters on the Stirling engine performance. De Boer [79] optimized a Stirling engine regenerative heat exchanger for a maximum possible value of the power output. Karabulut et al. [80] studied the improvement of the performance of beta-type Stirling engine. Three different displacers (without any surface treatment, zirconium coated with 0.15 mm thickness, and helically knurled with 0.30 mm track depth) were tested upon and the highest engine power was obtained with knurled displacer. Petrescu et al. [81] explained Stirling engine performance by considering the thermodynamic loss due to heat and pressure drop.

Sripakagorn and Srikam [82] developed a prototype Stirling engine working at the moderate temperature range. The performance of engine is evaluated at different values of charge pressures and wall temperatures at the heater section. Yaqi et al. [83] developed a mathematical model for the overall thermal efficiency of the solar-powered high temperature differential dish-Stirling engine with finite-rate heat transfer, regenerative heat losses, and conductive thermal bridging losses.

Krishnaiah et al. [84] presented an atlas of solar electricity potential of Stirling dish power generation system. They have developed maps of annual variation of solar electricity potentials, comparison of electricity potentials of Indian cities for different months, average electricity potentials and annual electrical energy generation for various Indian cities.

In central tower receiver solar thermal power plants (CTRSTPP), incident solar radiation is arranged to reflect from an array of large mirrors called heliostats and concentrated on a receiver situated at the top of a supporting tower [1]. A working fluid flowing through the receiver absorbs the concentrated radiation and transports the heat to the ground level where it is used to generate mechanical power through a thermodynamic power cycle like the Rankine or the Brayton cycle. Each heliostat at a central receiver facility has reflective surface area from 50 to 150 m^2, mirrors installed on a common pillar. The heliostats track the sun on two axes (east to west and up and down). There are different receiver classifications depending on the constructional configuration and the heat transfer medium. The geometrical configuration can be either external or cavity type. In a cavity receiver, the radiation reflected from the heliostats passes through the aperture into a box like structure before impinging on the heat transfer surface. External receivers can be designed with a flat-plate or cylindrically shaped tubular panels. This is the typical solution adopted for surround heliostats fields [85]. In volumetric receivers, air acts as a gaseous fluid typically operating from 373 to 1073 K. In 1986 under the initiative of SOTEL and DLR, the study of a 30 MW_e plant for Jordan was initiated. The international PHOEBUS Consortium was formed by companies from Germany, Switzerland, Spain, and the USA and the feasibility study completed in March 1990 [86]. The plant was successfully operated by DLR and CIEMAT for a total of nearly 400 h between April and December 1993, and for shorter periods in 1994 and 1999, demonstrating that a receiver outlet temperature of 973 K could easily be achieved within twenty minutes of plant start-up [87]. The performance data on central tower receiver thermal power plants for different receivers and heat carrying medium (molten salt and air) have been shown in Table 3. The brief review of the research works carried out on solar central receiver thermal power plants and its components are presented here for further reference.

Riaz [88] modeled solar concentrators of large area for central receiver power plants. Two governing factors like steering constraints on mirror orientations, and shadow effects by blocking the incident/reflected solar radiation are considered. Walzel et al. [89] presented the calculation of solar flux density on the central receiver due to a large number of flat polygonal reflectors having various orientations for the tower concept of solar energy collection. Peterka et al. [90] discussed mean and peak wind loads on flat rectangular or circular heliostats. Reduced wind loads were

Table 3
Performance data on central tower receiver thermal power plants [87].

Parameter	**Solar two** (Mature)	**Solar tres**	**Solar 50/Solar Cuatro**	**Solar 100**	**PS-10**
Working fluid	Molten salt	Molten salt	Molten salt	Molten salt	Air
Plant rating	10	15	50	100	10
Annual solar insolation (kWh/m^2)	2700	2067	2067	2700	2063
Capacity factor (%)	20	65	69	70	–
Field area (m^2)	81,400	263,000	971,000	1,466,000	89,271
Receiver thermal rating (MW)	42	120	466	796	55
Thermal storage size (MWh)	110	610	1850	3820	–
Steam generator rating (MW)	35	37	130	254	5.34
Annual net energy production (MWh)	16,600	75,500	302,000	613,000	19,200
Peak net efficiency	0.13	0.19	0.22	0.22	0.17
Annual net efficiency	0.08	0.14	0.15	0.16	0.12

demonstrated for heliostats within a field of heliostats and upper bound curves were developed to provide preliminary design coefficients. Ali [91] proposed a mathematical model to identify the starting time of the power plant, the height of the tower, the distance between the tower and the heliostat mirror, and the location of the power plant as the design parameters for Iraq. Buck et al. [92] designed and built a new secondary concentrator with improved efficiency for solar hybrid power plants. Several configurations of solar-hybrid gas turbine cycles in the low to medium power capacity range are examined for their performance and costs.

Schmitz et al. [93] demonstrated six types of heliostat field layouts for striving maximum efficiencies in solar thermal central receiver systems and possible potential improvement due to multiple apertures in central receiver systems with secondary concentrators. Sanchez and Romero [94] described the optimization procedure to calculate the yearly normalized energy surface available for a given tower height. Wu et al. [95] studied different gap sizes between the facets of the heliostats experimentally and numerically for the purpose of reduction of wind load on heliostats. Zoschak et al. [96] focused on the design and operating aspects of a 10-MW cavity-type, natural circulation steam generating receiver for a central receiver thermal power plant.

Wu et al. [97] described the conceptual design of an advanced water/steam receiver for a commercial-scale solar central receiver thermal power system. It consists of four separate cavities in a single receiver unit, each cavity receiving concentrated solar energy from one quadrant of a surrounding heliostat field. Carotenuto et al. [98] presented design of a prototype multi-cavity external flow air receiver and tested it at the Platform Solar de Almeria test facility. A good agreement between measured and predicted results was noticed, within the limits of accuracy of the data acquisition system of the facility and of the experimental system.

Eck et al. [99] presented the dual receiver concept for the improvement of the performance of the central receiver to the steam cycle in a solar thermal power plant. The water is evaporated directly in the tubular steam generator while preheating and superheating are done in heat exchangers by using the hot air from the volumetric receiver. The results confirm the benefits of the new concept for the annual mean efficiency which is increased from 13% to 16%. Buck et al. [100] studied the concept of open volumetric receiver technology for the PS-10 project in Spain.

Table 4
Geometrical and optical parameters for the collector loop considered [3,23,54].

Absorber tube outer diameter (m)	0.07
Absorber tube inner diameter (m)	0.065
Glass envelope outer diameter (m)	0.115
Glass envelope inner diameter (m)	0.109
Design point parameters (Jodhpur)	
Number of collectors	4
Solar beam radiation (W/m^2)	870
Longitude (deg)	73.017E
Latitude (deg)	26.28N
Incidence angle (N–S axis orientation) (deg)	12.03
Design point parameters (Delhi)	
Number of collectors	6
Solar beam radiation (W/m^2)	620
Longitude (deg)	77.18E
Latitude (deg)	28.57N
Incidence angle (N–S axis orientation) (deg)	9.48
Number of modules per collector	12
Width of the module (m)	5.76
Length of every module (m)	12.27
Mirror length in every module (m)	11.9
Ambient temperature (K)	304
Incidence angle (N–S axis orientation) (deg)	12.03
Focal length (m)	1.71
Drive	Hydraulic
Optical parameters for the collector	
Intercept factor (IF)	0.92
Mirror reflectivity, γ_r	0.92
Glass transmissivity, τ_g	0.945
Solar absorptivity, α_a	0.94
Peak optical efficiency, η_o	0.75
Thermal emissivity, ε_r	0.04795 +0.0002331 × T_r (°C)
Losses due to shading of heat collector element (HCE) by dust on the envelope, η_d	0.98

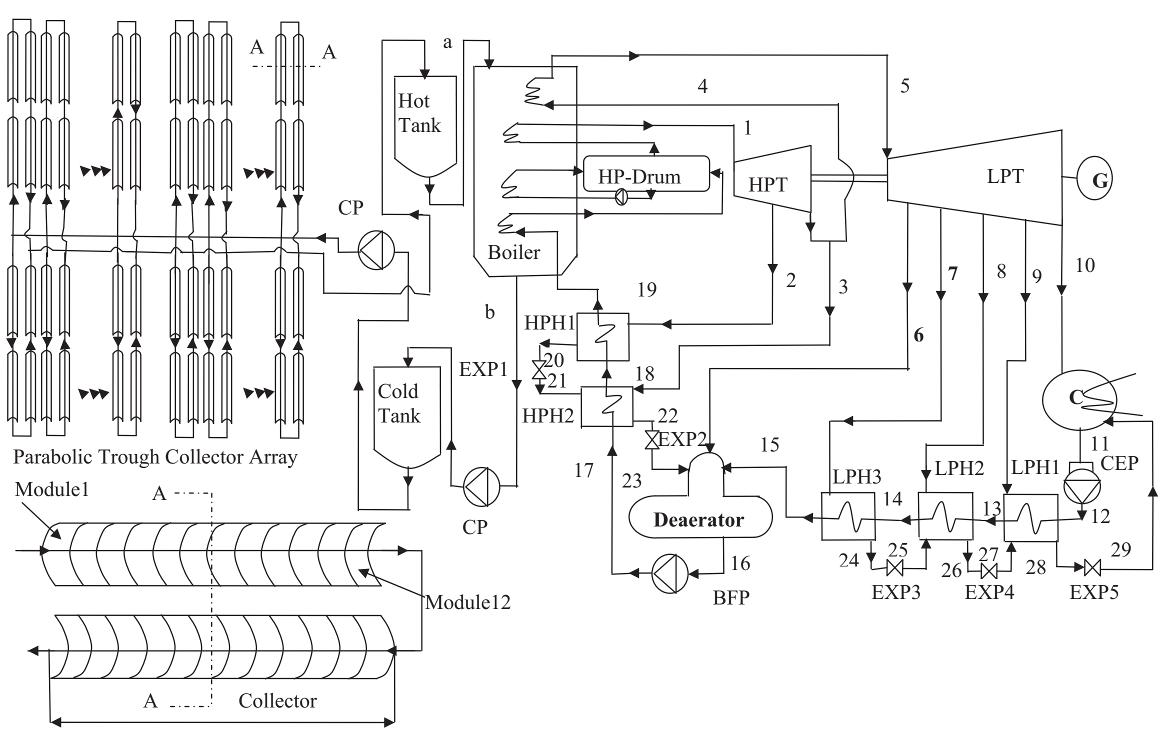


Fig. 2. Simplified schematic view of the 50 MW$_e$ parabolic trough concentrator solar thermal power plant [38].

Several benefits of the new concept; especially higher thermal efficiency of the receiver and the annual output increment by 27% as compared to the solar air heating system has been reported. Fang et al. [101] proposed a combined (Monte-Carlo) calculation method for evaluating the thermal performance of the solar cavity receiver. With this method, the thermal performance of a solar cavity receiver, a saturated steam receiver, is simulated under different wind environments. Antonio et al. [102] studied volumetric receivers and development of new designs to minimize heat losses and also discussed other important issues, such as the basic plant configuration, flow stability phenomenon and the main problems of a window design for pressurized receivers.

Yu et al. [103] presented the models of the collector and cavity receiver. The two models were coupled together based on the STAR-90 simulation platform. The results demonstrated that it can provide good control system design of the entire solar thermal power tower system. They also proposed an integrated receiver model for full range operation conditions in order to simulate and evaluate the dynamic characteristics of a solar cavity receiver. Based on this model [104], the dynamic characteristics of the solar cavity receiver were tested and also calculated thermal loss with different wind conditions. Montes et al. [105] analyzed a new optimized heat transfer model in the absorber surface of a thermofluidynamic design of a solar central receiver. This conceptual scheme has also been applied to the particular case of a molten salt single cavity receiver, although the configuration proposed is suitable for other receiver designs and working fluids.

Alvarez et al. [106] modeled a hybrid renewable power plant with two energy sources using a Mixed Logical Dynamical Modeling tool for the representation of hybrid systems. Collado [107] has given a simplified model for quick evaluations of the annual overall energy collected by a surrounding heliostat field. Yao et al. [108] presented modeling and simulation of DAHAN, the pioneer 1 MW central receiver using a software tool HFLD. Yang et al. [109] investigated the interaction between the heat transfer performance and the thermal efficiency of a molten salt receiver used in the solar power tower plant. The results of the experiment show that, by using the spiral tube as the heat transfer tube, the heat transfer performance of the molten salt receiver is obviously improved, and the radiation and convection losses are significantly reduced. Chacartegui et al. [110] proposed three different cycles, the first two of which are stand-alone closed cycle gas turbines using carbon dioxide and the third proposal is a combined cycle that comprises a topping carbon dioxide gas turbine and a bottoming organic Rankine cycle for central tower receiver solar thermal power plants. Leonardi and D'Aguanno [111] presented a new FORTRAN computer program for the simulation of the optical performance of a central receiver solar plant. The implemented mathematical algorithm allows for the calculation of cosine, shading and blocking effects for heliostats arbitrarily arranged in the solar field. Leonardi [112] presented a design of a beam-down solar power plant. The effect of the hyperboloid eccentricity on both the sun shape and the size of the heliostats were analyzed. Optimal values of the characteristic parameters of the compound parabolic concentrator are also calculated for yearly solar power collection.

Xu et al. [113] presented a theoretical framework for the energy and exergy analysis of the solar power tower system using molten salt as the heat transfer fluid. Both the energy losses and exergy losses in each component and in the overall system are evaluated to identify the causes and locations of the thermodynamic

Table 6
Year round comparative performance analysis results of a 50 MW_e PTCSTPP, for the tropical locations of Jodhpur and Delhi [38].

S. no.	Parameters for a year	**Jodhpur**	**Delhi**
1	Solar radiative energy input (MWh)	502,221.13	538,175.24
2	Useful thermal energy output of PTC (MWh_{th})	310,898.28	317,638.84
3	Overall year round efficiency of PTC (%)	**61.90**	**59.02**
6	Total electrical energy output per year (MWh_e)	113,602.42	115,715.82
7	Overall year round efficiency of plant (%)	**22.62**	**21.50**
8	Total collector field area (m^2)	**720,000**	**1,080,000**
9	Required land area for plant (ha)	**79.2**	**118.8**

Table 5
Stream data for 50 MW_e steam power cycle (at turbine inlet pressure of 105 bar and temperature of 643 K) [38].

S. ID	Fluid	Mass flow rate (kg/s)	Temperature (K)	Pressure (bar)	Sp. enthalpy (kJ/kg)	Sp. entropy (kJ/kg K)	Energetic power (kW)
1	Steam	65.52	643	105	2983	6.021	195,468
2	Steam	7.405	556.3	52.96	2854	6.061	21,131
3	Steam	4.961	494.9	24.03	2722	6.11	13,505
4	Steam	53.16	494.9	24.03	2722	6.11	144,708
5	Steam	53.16	643	21.21	3179	6.996	168,981
6	Steam	3.254	552.5	10.21	3006	7.033	9780
7	Steam	3.202	460.7	4.232	2833	7.086	9069
8	Steam	2.845	382.9	1.428	2659	7.155	7564
9	Steam	2.537	349.08	0.4038	2484	7.23	6301
10	Steam	41.32	315	0.08205	2296	7.326	94,858
11	Water	49.9	315	0.08205	175.8	0.5996	8771
12	Water	49.9	315.08	10.21	177.1	0.5995	8839
13	Water	49.9	346.72	10.21	309.4	0.9995	15,441
14	Water	49.9	380.5	10.21	451.3	1.39	22,523
15	Water	49.9	416.1	10.21	602.8	1.77	30,081
16	Water	65.52	451.8	10.21	757.9	2.128	49,660
17	Water	65.52	454.5	137.6	772.3	2.137	50,864
18	Water	65.52	491.8	137.6	941.5	2.486	61,691
19	Water	65.52	537.5	137.6	1155	2.901	75,700
20	Water	7.405	496.8	51.37	961.7	2.547	7121
21	Water	7.405	494.9	24.03	961.7	2.554	7121
22	Water	12.37	459.5	23.31	792.5	2.2	9799
23	Water	12.37	453.8	10.21	792.5	2.204	9799
24	Water	3.202	385.5	4.105	472	1.446	1511
25	Water	3.202	382.9	1.428	472	1.446	1511
26	Water	6.047	351.72	1.385	329.6	1.06	1933
27	Water	6.047	349.08	0.4038	329.6	1.06	1933
28	Water	8.583	320.08	0.3917	197.2	0.6527	1692
29	Water	8.583	314.97	0.08205	197.2	0.6534	1692

imperfection. The results show that the maximum exergy loss occurs in the receiver system, followed by the heliostat field system, although main energy loss occurs in the condenser of the power cycle system. They also presented model of 1 MW Dahan solar thermal power tower plant using mathematical modular modeling method. The dynamic and static characteristics of the power plant are analyzed based on these models [114].

From the above reported findings, the technical viability and reliability of the medium and high temperature solar thermal power plants is proved. Another most important issue for commercialization of the technologies is the system cost. Reported installation costs of PDCSSPP are very high, i.e., approximately $10,000 per kW. However, it is estimated that the cost of the system may reduce to $2500 per kW, if there is installations of more units (i.e., 500 units per year). The installation cost of CTRSTPP is also reported higher, i.e. $ 14,000 per kW [1]. In the present paper, the authors have tried to highlight the case studied on economical feasibility of the PTCSTPP, PDCSSPP, and CTRSTPP solar thermal power plants for the locations of Jodhpur and Delhi in India [123] with reference to published works of the authors elsewhere. It will be very helpful in further design and development of solar thermal power plants anywhere in the world, especially in India.

4. Economic assessment of low, medium and high temperature solar thermal power plants for Indian tropical climates—case studies

Study of the year round performance of low, medium and high temperature solar thermal power plants for Indian tropical climates is scant in literature for determining the unit cost of solar thermal power generation. In the present study, economical assessments of the solar thermal power generation option based on different concentration technologies have been done. A reasonable capacity of 50 MW has been considered at two selective tropical locations of India, i.e., Jodhpur and Delhi. The DNI, ambient temperature and wind velocities for selected Indian locations are collected from EERE [115] website.

These costs have been estimated according to system advisor model [116] and the dollar is converted into Indian rupees ($1= INR 50 as on November 10, 2011).

First of all the annual electricity production is estimated in each case, then economic analysis is performed to calculate the cost of the unit (kWh_e) electric energy generation cost (UC), which can be used to compare different locations [117,118].

$$UC = \frac{CRF \times C_{IN} + C_{OM} + C_F}{E_{NET}} \quad (1)$$

where CRF is uniform series capital recovery factor [119].

$$CRF = \left[\frac{i(1+i)^n}{(1+i)^n - 1}\right] \quad (2)$$

where C_{IN} is installation cost, C_{OM} is operation & maintenance cost, and C_F is fuel cost. In this analysis, taxes, incentives, and insurance are not considered. Interest rate (i) (%) and lifespan (n) of a power plant (years) are taken into account.

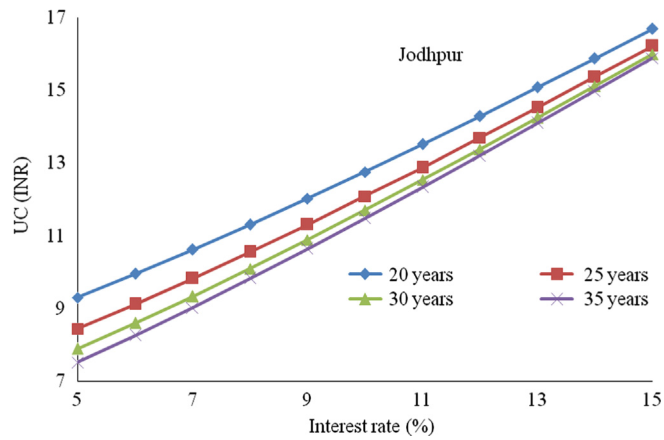


Fig.3. Variation of unit (kWh_e) electric energy generation cost (UC) with interest rate (%) of PTCSTPP for the Jodhpur location [123].

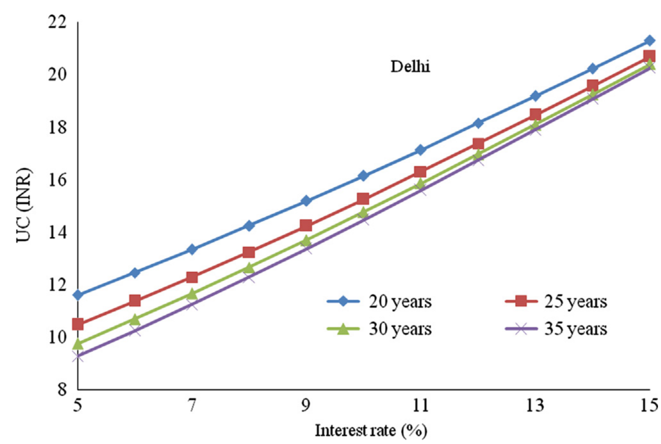


Fig. 4. Variation of unit (kWh_e) electric energy generation cost (UC) with interest rate (%) of PTCSTPP for the Delhi location [123].

Table 7
Cost data for economical analysis of 50 MW_e PTCSTPP, for the tropical locations of Jodhpur and Delhi [123].

Total product cost	Cost [116]	Jodhpur	Delhi	Total cost (Jodhpur)	Total cost (Delhi)
Direct cost				INR (Lakhs)	INR (Lakhs)
Site improvements	150 INR/m^2	792,000 m^2	1,188,000 m^2	9900.00	14,850.00
Cost of collector	19,250 INR/m^2	276,480 m^2	414,720 m^2	53,222.40	79,833.60
Storage	4000 INR/kWh_t	110,000 kWh_t	110,000 kWh_t	4400.00	4400.00
Power plant	47,000 INR/kW_e	55,000 kW_e	55,000 kW_e	25,850.00	25,850.00
Total direct cost				93,372.40	124,933.60
Indirect cost					
Procure, construction and execution	11% direct cost	9,337,240,000	12,493,360,000	10,270.96	13,742.70
Project, land, miscellaneous	2% direct cost	9,337,240,000	12,493,360,000	1867.45	2498.67
Total indirect cost				12,138.41	16,241.37
Total (direct +indirect) cost				105,510.81	141,174.97
Operation and maintenance costs					
Fixed cost by capacity	3500 INR/kW/year	55,000 kW/year	55,000 kW/year	1925.00	1925.00
Variable cost by generation	150 INR/MWh_e	113,602.42 MWh_e	115,715.82 MWh_e	170.40	173.57
Total (O&M) cost				2095.40	2098.57

4.1. Techno-economic analysis of PTCSTPP

The solar thermal power system reported in the present study has 80 loops of the parabolic trough collector array oriented N–S axis and E–W operating in a tracking mode [38]. The collector–receiver subsystem and Rankine heat engine subsystem are shown in Fig. 2. Therminol VP-1 oil at 566 K is pumped from a 'cold' storage tank through the receiver where it is heated to 643 K and then on to a 'hot' tank for storage. The design parameters of the parabolic collector loop are given in Table 4. The property data at the various stream state points of a 50 MW_e steam power plant cycle are shown in Table 5 having the HPT inlet pressure of 105 bar and temperature of 643 K.

Based on the performance analysis carried out by Siva Reddy et al. [38], a detailed comparative performance result of 50 MW_e PTCSTPP for the locations of Jodhpur and Delhi is presented in Table 6. For Jodhpur location with 79.2 ha land area and having solar radiative energy input of 502.2×10^3 MWh per annum, 113.6×10^3 MWh_e power production has been estimated. In the similar fashion for Delhi location, with 118.8 ha land area and having solar radiative energy input of 538×10^3 MWh per annum, 115.7×10^3 MWh_e power production has been estimated. Performance efficiency of the PTCSTPP at the location of Jodhpur and Delhi is found to be 22.62%, 21.50% respectively.

Data for economic analysis are shown in Table 7. These costs have been evaluated according to System Advisor Model [116] from National Renewable Energy Laboratory. Eqs. (1) and (2) are used for determining unit electrical energy generation cost. The installation and operation and maintenance cost of the PTCSTPP according to the selected Indian tropical climates like Jodhpur and Delhi have been estimated.

The installation cost per MW_e electrical capacity is INR 21.10 crore for Jodhpur and INR 28.23 crore for Delhi based on the solar intensity availability. INR 41.91 lakhs and INR 41.97 lakhs as operation and maintenance cost per MW_e electrical capacity for the locations of Jodhpur and Delhi respectively. Variation of unit (kWh_e) electric energy generation cost (UC) with interest rate (%) for different lifespan of a power plant at Jodhpur and Delhi has been shown in Figs. 3 and 4. Unit cost is varying from INR 7 to 17 and INR 9 to 22 with interest rate for different lifespan of the PTCSTPP for the locations of Jodhpur and Delhi. For different lifespan of power plant, unit (kWh_e) electric energy generation cost increases linearly with respect to the interest rate (%). The effect of the interest rate is more as compared to the lifespan of the power plant. At general condition (30 years life span of the plant and 10% interest rate) unit (kWh_e) electric energy generating cost is obtained as INR 11.70 and INR 14.76 respectably for the location Jodhpur and Delhi.

4.2. Techno-economic analysis of PDCSSPP

The PDCSSPP system considered here consists of a parabolic dish concentrator, receiver and Stirling engine as illustrated in Fig. 5. There is a dual axis tracking system with parabolic dish mirror to concentrate solar radiation on a receiver which is an integral part of Stirling engine. Heat pipe absorbers are used to transfer available heat energy into it. In the present analysis, designed capacity of 50 MW_e is considered in which the parabolic dish system of 25 kW_e is arranged in a matrix form like 50×40 of 2000 units of the design capacity resulting 50 MW_e. Three stars in Fig. 5 depict that there is a series of parabolic dish concentrating solar Stirling engines in between which cannot be shown. The designed data for 25 kW_e is shown in Tables 8 and 9.

The performance of the PDCSSPP has given better efficiency in Jodhpur as compared to Delhi, due to high DNI availability. Table 10 shows detailed comparative performance results of 50 MW_e PDCSSPP for the locations of Jodhpur and Delhi. For the Jodhpur location with 98.32 ha land area, it can produce 95.77×10^3 MWh_e at a solar radiative energy input of 384.06×10^3 MWh per annum. For the Delhi location with 122.74 ha land area, it can produce 86.21×10^3 MWh_e at a solar radiative energy input 347.61×10^3 MWh per annum.

Data for economic analysis are shown in Table 11. These costs have been carefully taken according to System Advisor Model (2011). INR13 and INR 15 crore as installation cost per MW_e electrical capacity for the location Jodhpur and Delhi, INR 28 lakhs

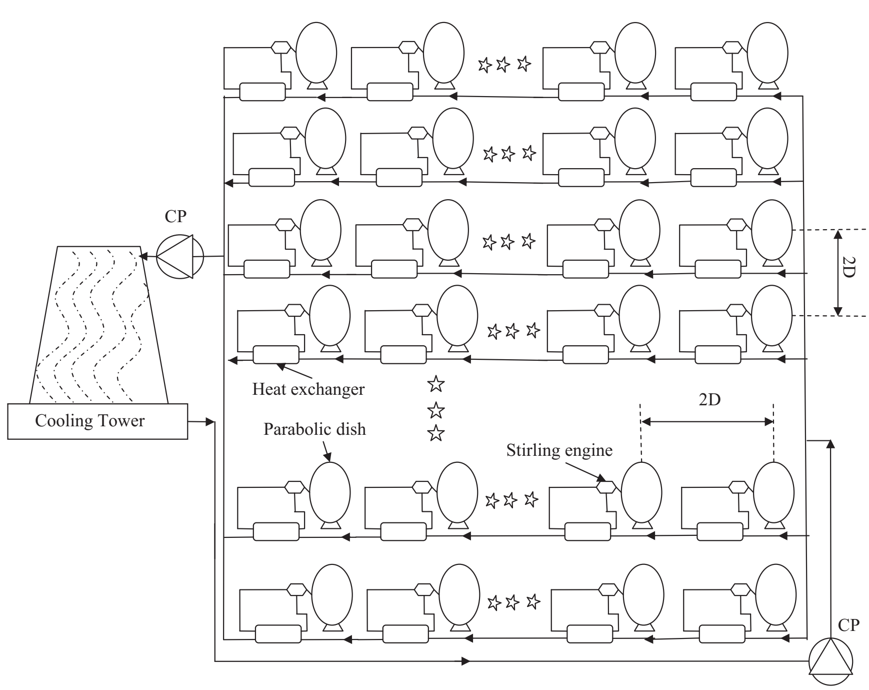


Fig. 5. Simplified schematic view of the 50 MW_e parabolic dish concentrating solar Striling engine power plant [120].

as operation and maintenance cost per MW_e electrical capacity for thought out the year the year. Variation of unit (kWh_e) electric energy generation cost (UC) with interest rate (%) for different lifespan of a solar power plant, for the locations Jodhpur and Delhi have been shown in Figs.6 and 7 respectively. Unit cost is varying from INR 5 to INR 12 for the location Jodhpur and INR 7 to INR 16 for the location Delhi with respect to interest rate. In both locations for different lifespan of power plant, unit (kWh_e) electric energy generation cost increases linearly with respect to interest rate (%). At general condition (30 years life span of the plant and 10% interest rate) unit (kWh_e) electric energy generating cost obtained as INR 8.76 and INR 11.06 for the locations, Jodhpur and Delhi respectively.

4.3. Techno-economic analysis of CTRSTPP

In this case study heliostat field-receiver subsystem as shown in Fig. 8 consists of heliostats and volumetric air receiver at the focal point on the central tower. Heliostats array consists of 2545 collectors for Jodhpur and 3530 collectors for Delhi based on the DNI availability to obtain design operating temperature of air (973 K). Each heliostat module is made up of 10.5×11 m^2 dimension with an effective reflector area of 100 m^2. The height of the central tower receiver has been considered as 150 m. The various design parameters of the central tower receiver are given in Table 12. The Rankine heat engine model is considered for single reheating for avoiding dryness fraction in the last stage of low pressure turbine. Table 13 shows the property data of the stream at different state points for a 50 MW_e steam power plant cycle at the HPT inlet pressure and temperature 185 bar and 813 K respectively.

Table 14 shows detailed comparative performance results of 50 MW_e CTRSTPP for the locations of Jodhpur and Delhi. For the Jodhpur location with 146.3 ha land area, it can produce 123.07×10^3 MWh_e at a solar radiative energy input of 490.76×10^3 MWh per year. For the Delhi location with 257.3 ha land area, it can produce 124.88×10^3 MWh_e at a solar radiative energy input 487.43×10^3 MWh per year.

The simple formulation for unit (kWh_e) electric energy generation cost (UC) was as given in Eqs. (1) and (2). Input/Output data for economic analysis is shown in Table 15. The installation, operation & maintenance cost of the CTRSTPP according to the selected Indian tropical climates like Jodhpur and Delhi has been estimated. The installation cost per MW_e electrical capacity is found to be INR 23.56 crore for Jodhpur and INR 28.40 crore for Delhi based on the solar intensity availability. While the operation and maintenance costs INR 42.19 lakhs and INR 42.25 lakhs per MW_e electrical capacity for the locations of Jodhpur and Delhi respectively. Variation

Table 8
Design characteristics of a solar parabolic dish-Stirling engine (25 kW_e) [121,122].

Concentrator	
Aperture diameter (m)	10.57
Aperture area (m^2)	91.01
Glass area (m^2)	87.67
Focal length (m)	7.45
Rim angle (deg)	39
Intercept factor, IF	0.92
Mirror reflectivity, γ_r	0.92
Cavity absorptivity, α_c	0.94
Losses due dust on the envelope, η_d	0.98
Cavity emissivity, ε_c	0.90
Module dimensions	11.89 m H, 11.28 m W
Module weight (kg)	6.934
Tracking	Azimuth/elevation
Cavity diameter (mm)	450 (Inconel 625 material)
Absorber diameter (mm)	200
Gas operating temperature (K)	1033
Max tube temperature (K)	1083
Stirling engine (kinematic)	
Engine dry weight (kg)	225
Displacement volume (cm^3)	4×95
Bore and stroke (mm)	55 and 40
Regenerators	4×2@44 mm long, 57 mm diameter, holding 200 mesh stainless steel wire screens
Number of pistons	4. double acting
Working fluid	H_2 or He
Working fluid pressure (max) (MPa)	20
Operating temperature (°C)	720
Power control	Fluid pressure
Cooling	Water
Output power	27 kW (max), 22 kW (rated)
Power conversion unit	
Alternator	Induction, 1800 rpm
Alternator efficiency (%)	92–94
Electric energy	480 V, 60 Hz, three phase
Gross power rating (kW)	25 at 1000 W/m^2
Minimum solar insolation (W/m^2)	250–300

Table 10
Year round comparative performance analysis results of a 50 MW_e PDCSSPP, for the tropical locations of Jodhpur and Delhi [120,123].

S. no.	Parameters for a year	**Jodhpur**	**Delhi**
1	Solar radiative energy input (MWh)	384,062.94	347,608.37
2	Useful thermal energy output of PDC (MWh_{th})	292,880.33	265,225.68
3	Overall year round efficiency of PDC (%)	76.26	76.03
4	Electric energy output (MWh_e)	95,775.3	86,206.72
5	Overall year round efficiency of plant (%)	24.93	24.80
6	Total collector field area (m^2)	893,800	1,115,809
7	Required land area for plant (ha)	98.32	122.74

Table 9
Design characteristics of a PDCSSPP (50 MW_e) for Jodhpur and Delhi [120,123].

	Jodhpur	**Delhi**
Design point parameters		
Number of parabolic dish systems	2000	2000
Gross power rating of each parabolic dish system (kW)	25	25
Longitude (deg)	73.017E	77.18E
Latitude (deg)	26.28N	28.57N
Designed solar irradiance (W/m^2)	1000 (DNI)	800 (DNI)
Power consumption for cooling tower system, power needed for the tracking	$0.03 \times$ output power	$0.03 \times$ output power
Concentrator		
Aperture diameter (m)	10.57	11.81
Aperture area (m^2)	91.01	113.81
Glass area (m^2)	87.67	109.57

Table 11
Cost data for economical analysis of 50 MW_e PDCSSPP, for the tropical locations of Jodhpur and Delhi [120,123].

Total product cost	Cost [116]	Jodhpur	Delhi	Total cost (Jodhpur)	Total cost (Delhi)
Direct cost				INR (Lakhs)	INR (Lakhs)
Site improvements	150 INR/m^2	893,800 m^2	1,115,809 m^2	1340.70	1673.71
Cost of collector	20,000 INR/m^2	182,020 m^2	227,620 m^2	36,404.00	45,524.00
Receiver cost	12500 INR/kW	55,000 kW	55,000 kW	6875.00	6875.00
Engine cost	25,000 INR/kW	55,000 kW	55,000 kW	13,750.00	13,750.00
Total direct cost				58,369.70	67,822.71
Indirect cost					
Procure, construction & execution	11% Direct cost	5,836,969,880	6,782,271,320	6420.67	7460.50
Project, land, miscellaneous	2% Direct cost	5,836,969,880	6,782,271,320	1167.39	1356.45
Total indirect cost				7588.06	8816.95
Total (direct +indirect) cost				65,957.76	76,639.67
Operation and maintenance costs					
Fixed cost by capacity	2500 INR/kW/year	55,000 kW/year	55,000 kW/year	1375.00	1375.00
Variable cost by generation	35 INR/MWh_e	95,775.3 MWh_e	86,206.7 MWh_e	33.52	30.17
Total (O&M) cost				1408.52	1405.17

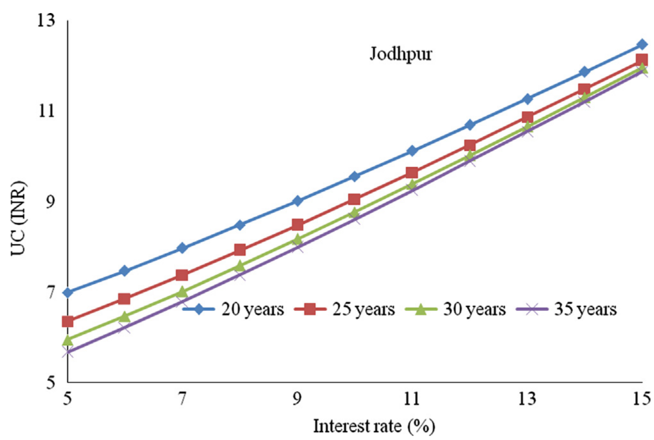


Fig. 6. Variation of unit (kWh_e) electric energy generation cost (UC) with interest rate (%) of PDCSSPP for the Jodhpur location [120].

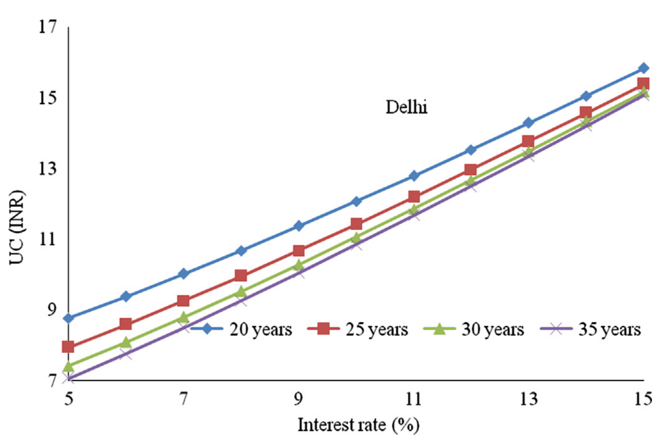


Fig. 7. Variation of unit (kWh_e) electric energy generation cost (UC) with interest rate (%) of PDCSSPP for the Delhi location [123].

of unit (kWh_e) electric energy generation cost (UC) with interest rate (%) for different lifespan of a power plant at Jodhpur and Delhi has been shown in Figs. 9 and 10. Unit cost is varying from INR 5 to 16 and INR 7 to 19 with interest rate for different lifespan of the CTRSTPP for the locations of Jodhpur and Delhi. For different lifespan of power plant, unit (kWh_e) electric energy generation cost increases linearly with respect to the interest rate (%). The effect of the interest rate is

Table 12
Geometrical and optical parameters for the heliostat field considered [86].

Heliostat dimension (m × m)	10.5 × 11
Reflective area per generic heliostat (m × m)	10 × 10
Incident power (MW_t)	197
Receiver elevation (above ground) (m)	150
Receiver shape	Half cylinder
Tilt of absorber plane (deg)	15
Receiver rated output (MW_t)	134
Exit air temperature (K)	973
Return air temperature (K)	473
Design point parameters (Jodhpur)	
Number of heliostat	2545
Solar beam radiation (W/m^2)	870
Longitude (deg)	73.017E
Latitude (deg)	26.28N
Design point parameters (Delhi)	
Number of heliostat	3530
Solar beam radiation (W/m^2)	620
Longitude (deg)	77.18E
Latitude (deg)	28.57N
Tracking	Azimuth/elevation
Optical parameters for the collector	
Intercept factor, IF	0.92
Mirror reflectivity, γ_r	0.92
Cavity absorptivity, α_c	0.94
Losses due dust on the envelope, η_d	0.98
Cavity emissivity, ε_c	0.90

more as compared to the lifespan of the power plant. At general base condition (30 years life span of the solar plant and 10% interest rate), the unit (kWh_e) electric energy generating cost is obtained as INR 10.09 and INR 12.10 for the location of Jodhpur and Delhi respectively.

5. Conclusions

Based on the present literature review, the authors conclude that there is no doubt in the technical feasibility of solar thermal power plants for commercialization in the present scenario. There is a need to substitute the demand of power with solar energy to reduce the rate of consumption of fossil fuels and consequently reducing green house gas emissions. The performance and economic analysis carried out for the solar thermal power plants (PTCSTPP, PDCSSPP, and CTRSTPP) for the locations of Jodhpur and Delhi to explore the possibility of solar thermal power generation in India is presented here. The estimated unit (kWh_e) electric

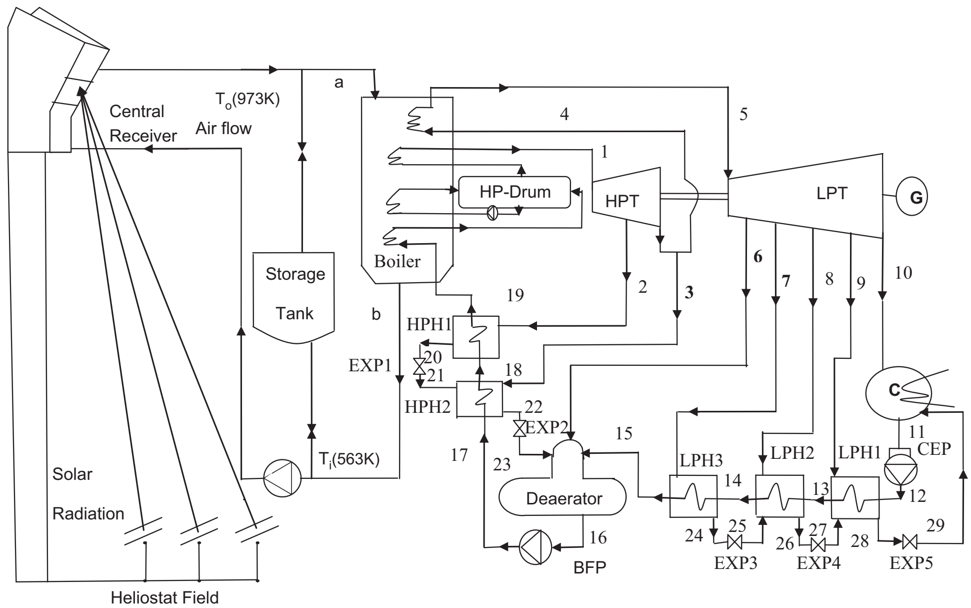


Fig. 8. Simplified schematic view of the 50 MW_e central tower receiver solar thermal power plant [123].

Table 13
Stream data for 50 MW_e Steam power cycle (at turbine inlet pressure of 185 bar and temperature of 813 K) [123].

S. ID	Fluid	Mass flow (kg/s)	Temperature (K)	Pressure (bar)	Sp. enthalpy (kJ/kg)	Sp. entropy (kJ/kg K)	Energetic power (kW)
1	Steam	55.63	813	185	3382	6.352	188,126
2	Steam	6.275	705.4	93.31	3204	6.395	20,102
3	Steam	4.851	611.5	46.25	3047	6.447	14,780
4	Steam	44.5	611.5	46.25	3047	6.447	135,585
5	Steam	44.5	753	40.82	3398	7.02	151,245
6	Steam	2.978	637.8	17.98	3173	7.062	9452
7	Steam	2.916	533.5	7.456	2974	7.115	8672
8	Steam	2.54	429.1	2.516	2777	7.196	7054
9	Steam	2.884	363.36	0.712	2590	7.281	7471
10	Steam	33.19	315	0.082	2325	7.42	77,165
11	Water	41.53	315	0.082	175.8	0.598	7298
12	Water	41.53	315.17	17.98	178.2	0.601	7398
13	Water	41.53	360.92	17.98	369.5	1.167	15,345
14	Water	41.53	398.1	17.98	526.7	1.581	21,871
15	Water	41.53	437.8	17.98	697.1	1.989	28,947
16	Water	55.63	477.8	17.98	874.2	2.376	48,630
17	Water	55.63	482.6	214.6	903.4	2.39	50,258
18	Water	55.63	528.8	214.6	1114	2.805	61,946
19	Water	55.63	575.7	214.6	1347	3.227	74,910
20	Water	6.275	533.8	90.98	1138	2.879	7139
21	Water	6.275	532.1	46.25	1138	2.89	7139
22	Water	11.13	487.6	45.09	919.5	2.463	10,231
23	Water	11.13	480.1	44.86	919.5	2.47	10,231
24	Water	2.916	403.1	7.23	547.3	1.636	1596
25	Water	2.916	400.6	7.23	547.3	1.637	1596
26	Water	5.456	365.92	2.44	389.4	1.226	2124
27	Water	5.456	363.36	0.712	389.4	1.227	2124
28	Water	8.34	320.17	0.687	197.5	0.667	1648
29	Water	8.34	314.97	0.082	197.5	0.668	1648

Table 14
Year round comparative performance analysis results of a 50 MW_e CTRSTPP, for the tropical locations of Jodhpur and Delhi [123].

S. no.	Parameters for a year	**Jodhpur**	**Delhi**
1	Solar radiative energy input (MWh)	49,0759.6	48,7434.52
2	Useful thermal energy output of CTR (MWh_{th})	310,262.67	314,846.76
3	Overall year round thermal efficiency of CTR (%)	**63.22**	**64.59**
4	Total electric energy output per year (MWh_e)	123,066.44	124,884.73
5	Overall year round efficiency of plant	**25.08**	**25.62**
6	Total collector field area (m^2)	**1,323,966**	**2,340,171**
7	Required land area for plant (ha)	**146.3**	**257.3**

Table 15
Cost data for economical analysis of 50 MW_e CTRSTPP, for the tropical locations of Jodhpur and Delhi [123].

Total product cost	Cost [86,116]	Jodhpur	Delhi	Total cost (Jodhpur)	Total cost (Delhi)
Direct cost				INR (Lakhs)	INR (Lakhs)
Site improvements	150 INR/m^2	1,456,362.6 m^2	2,574,188.1 m^2	2184.54	3861.28
Cost of collector	20,000 INR/m^2	254,500 m^2	353,000 m^2	50,900.00	70,600.00
Receiver cost	42,000 INR/kW_e	55,000	55,000	23,100.00	23,100.00
Storage	1500 INR/kWh_t	110,000 kWh_t	110,000 kWh_t	1650.00	1650.00
Tower cost	400,000 INR/m	150 m	150 m	600.00	600.00
Power plant	47,000 INR/kW_e	55,000 kW_e	55,000 kW_e	25,850.00	25,850.00
Total direct cost				104,284.54	125,661.28
Indirect cost					
Procure, construction and execution	11% Direct cost	10,428,454,390	12,566,128,215	11,471.30	13,822.74
Project, land, miscellaneous	2% Direct cost	10,428,454,390	12,566,128,215	2085.69	2513.23
Total indirect cost				13,556.99	16,335.97
Total (direct +indirect) cost				117,841.53	141,997.25
Per MW installation cost				**2356.83**	**2839.94**
Operation and maintenance costs					
Fixed cost by capacity	3500 INR/kW/year	55000 kW/year	55,000 kW/year	1925.00	1925.00
Variable cost by generation	150 INR/MWh_e	123,066.44 MWh_e	124,884.73 MWh_e	184.60	187.33
Total (O&M) cost				2109.60	2112.33
Per MW O&M cost				**42.19**	**42.25**

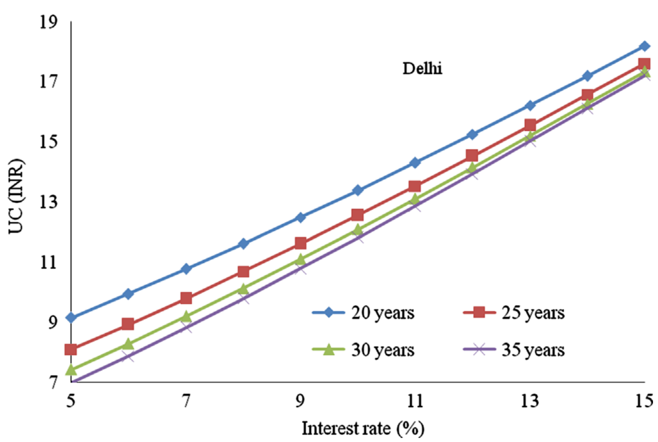


Fig. 9. Variation of unit (kWh_e) electric energy generation cost (UC) with interest rate (%) of CTRSTPP for the Jodhpur location [123].

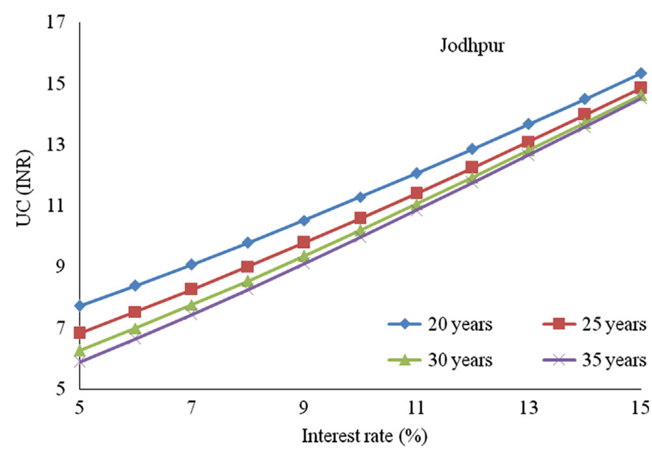


Fig. 10. Variation of unit (kWh_e) electric energy generation cost (UC) with interest rate (%) of CTRSTPP for the Delhi location [123].

energy generating cost for PTCSTPP is found to be INR 11.70 and INR 14.76. While unit (kWh_e) electric energy generating cost for PDCSSPP is estimated as INR 8.76 and INR 11.06 and the same for CTRSTPP is INR 10.09 and INR 12.10 for the locations Jodhpur and Delhi with the same lifespan (30 years) of the solar power plants and interest rate (10%) on investment. The unit power generation cost of the PDCSSPP is less than the PTCSTPP and CTRSTPP because its efficiency is high. However, the year round power output from the PDCSSPP is less compared to PTCSTPP and CTRSTPP; the main reason for this is that the PTCSTPP and CTRSTPP systems have thermal storage facility. PDCSSPP system is required to be designed at maximum DNI availability, because of no storage facility available to the system itself. Fluctuation in power output will take place with this system.

Above findings may motivate the researchers and policy makers to design and develop suitable solar thermal power plants for Indian climatic conditions based on the experience gained from the other part of the world. If the energy losses at various locations, such as production, transmission and distributions from power produced through conventional thermal power plants and environmental degradation impacts is taken into account, the cost of solar power may become competitive very soon. It is observed that the solar thermal power plants have come out of the experimental stage to commercial applications.

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